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Analysis of Homogeneous Charge Compression Ignition (HCCI) Engines for Cogeneration Applications

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Abstract

This paper presents an evaluation of the applicability of Homogeneous Charge Compression Ignition Engines (HCCI) for small-scale cogeneration (less than 1 MWe) in comparison to five previously analyzed prime movers. The five comparator prime movers include stoichiometric spark-ignited (SI) engines, lean burn SI engines, diesel engines, microturbines and fuel cells. The investigated option, HCCI engines, is a relatively new type of engine that has some fundamental differences with respect to other prime movers. Here, the prime movers are compared by calculating electric and heating efficiency, fuel consumption, nitrogen oxide (NO_x) emissions and capital and fuel cost. Two cases are analyzed. In Case 1, the cogeneration facility requires combined power and heating. In Case 2, the requirement is for power and chilling.

The results show that the HCCI engines closely approach the very high fuel utilization efficiency of diesel engines without the high emissions of NO_x and the expensive diesel fuel. HCCI engines offer a new alternative for cogeneration that provides a unique combination of low cost, high efficiency, low emissions and flexibility in operating temperatures that can be optimally tuned for cogeneration systems. HCCI engines are the most efficient technology that meets the oncoming 2007 CARB NO_x standards for cogeneration engines. The HCCI engine appears to be a good option for cogeneration systems and merits more detailed analysis and experimental demonstration.

Nomenclature

<u>Brake mean effective pressure (BMEP):</u> Specific engine power per unit of volumetric displacement, calculated as the work done in one engine cycle divided by the engine displacement.
<u>Brake thermal efficiency:</u> mechanical work produced by the engine divided by the lower heating value of the fuel consumed.
<u>CARB:</u> California Air Resources Board.
<u>Chilling efficiency:</u> chilling power divided by the lower heating value of the fuel consumed.

Electric efficiency: electric power produced by the engine-generator divided by the lower heating value of the fuel consumed.
Heating efficiency: Recovered thermal energy for cogeneration divided by the lower heating value of the fuel consumed.
Fuel utilization efficiency: For combined power and heating (Case 1) is the sum of the electric efficiency and heating efficiency. For combined power and chilling (Case 2) is the sum of the electric efficiency and the chilling efficiency.

Introduction

Cogeneration has the potential to considerably increase the fuel utilization efficiency of a prime mover from ~30% (if only the mechanical power is used) to ~70% (if the waste heat is used along with the mechanical power). However, cogeneration has achieved extremely small market penetration despite its great potential as an energy saving technology [1]. Reasons commonly cited for the low market penetration include high initial cost, need for frequent maintenance, high environmental emissions and utility policies that discourage cogeneration [2]. New cogeneration technologies may be able to bring this energy saving process into widespread use.

Cogeneration in small facilities (under 1 MW of electric power) can be accomplished with a variety of prime movers. These include spark-ignited (SI) engines, diesel engines, microturbines and fuel cells. While these options can be successfully applied for cogeneration, they all have some significant challenges with respect to cost, maintenance, emissions, or efficiency.

Stoichiometric SI engines can be equipped with a three-way catalyst to make them clean enough to meet even the most strict emissions standards, such as the current CARB standards for cogeneration engines (0.32 g/kWh NO_x) and the oncoming 2007 CARB standards (0.032 g/kWh NO_x)[1]. However, SI engines have low brake thermal efficiency (up to 35%[2]), because the compression ratio has to be kept low (~10:1) to avoid knock [1]. Knock also limits the air intake temperature and the cooling water temperature into the engine, which reduces the quality and quantity of the waste heat recovery that can be used for cogeneration. Stoichiometric SI engines also require high maintenance, due to the need to replace spark plugs and the catalytic converter.

Lean burn spark-ignited (SI) engines (those running with excess air) have a reasonably good brake thermal efficiency (up to 38%) due to the improved thermodynamic properties of the working fluid (high specific heat ratio, γ) [3]. However, lean SI engines are major sources of nitrogen oxide (NO_x) emissions. A recent publication [4] has

[1] While currently there are no federal (EPA) emissions standards in the US for stationary engines, CARB standards are considered to be representative of current and oncoming standards that may exist in urban areas and impacted basins.
[2] Efficiencies listed in this paper refer to engines with a displacement of 2 liters per cylinder, which are typical of ~200 kW engines. Higher engine efficiency can be obtained in bigger engines due to reduced heat transfer losses.

established that lean SI engines have intrinsically high NO_x emissions (0.6 g/kWh or higher), and there is no practical way to control these emissions, since lean NO_x aftertreatment technologies (i.e. urea selective catalytic reduction, SCR) are not considered economically viable for units under 1 MW. The lower bound of NO_x emissions from lean SI engines (0.6 g/kWh) is higher than the current CARB standard (0.32 g/kWh), limiting the applicability of these engines to areas with less strict emissions standards.

The diesel engine is the most efficient prime mover available for small facilities. Typical brake thermal efficiencies for diesel engines are over 40%. Diesel engines are relatively inexpensive and have low maintenance requirements. However, diesel engines also have intrinsically high emissions of NO_x. The lower bound for NO_x emissions from a diesel engine with no aftertreatment has been set at 1.3 g/kWh [4]. Again, there is no viable NO_x aftertreatment technology for diesel engines of small size.

Microturbines have a high temperature exhaust gas stream that can be efficiently applied for cogeneration. However, microturbines have low brake thermal efficiency (~25%) and high cost (~$1400/kWh). Microturbine NO_x emissions are low, but they may not be low enough to meet the very strict CARB 2007 emissions standards.

Fuel cells are a promising cogeneration technology, and phosphoric acid fuel cells have been commercialized with good efficiency (40%) and good potential for cogeneration [5]. Long durability and low maintenance have been successfully demonstrated [6]. The main obstacle to fuel cell commercialization is their cost (~$3000/kW) that limits their applicability to facilities that require very high quality power (e.g. banks, hospitals and computer facilities).

Homogeneous charge compression ignition (HCCI) engines have recently emerged as an alternative prime mover for transportation and stationary applications. Automobile and engine manufacturers are interested in HCCI engines due to their potential for high efficiency and low emissions. HCCI combustion is fundamentally different than combustion in spark-ignited (SI) engines and diesel engines. HCCI combustion is a thermal autoignition of a premixed fuel-air mixture, with no flame propagation (as in SI engines) or mixing-controlled combustion (as in diesel engines) [7]. HCCI engines can run extremely lean (equivalence ratio ~0.4) or very dilute (residual gas fraction ~0.6). In either case, the combustion temperature is low enough that the engine produces extremely low NO_x emissions (below the CARB 2007 standards) with no need for aftertreatment. Unlike SI engines, HCCI engines are not limited to low intake temperature, low cooling water temperature or low compression ratio due to knock. In fact, HCCI engines require a high intake temperature for ignition to occur, and HCCI engines may be described as a continuously knocking engine where the harmful effects of knock have been avoided by keeping a low combustion temperature [8]. HCCI engines can therefore operate at high compression ratios and high cooling water and air intake temperatures, improving the brake thermal efficiency as well as the quantity and quality of waste heat that can be successfully recovered. HCCI engines are intrinsically flexible in operating over a broad range of delivery and return water temperatures. This flexibility is important for applications that require high quality waste heat, such as absorption chillers. Finally, HCCI engines do not require spark plugs or a three-way catalyst, and are therefore expected to have lower maintenance costs than SI engines. Knock in HCCI engines does not result in increased maintenance or shortened engine life as long as the combustion temperature is kept low and the maximum peak cylinder pressure and rate of pressure rise specified by the manufacturer are not exceeded. In summary, HCCI engines have a combination of characteristics that make them desirable for cogeneration, including high brake and heating efficiency, low NO_x emissions, low cost and low maintenance requirements.

HCCI engines do present some technical challenges that have so far kept them from widespread commercialization. The main hurdles are combustion timing control, low specific power output, high emissions of hydrocarbon (HC) and carbon monoxide (CO), and difficulty to start when cold [7]. These are formidable technical challenges for transportation applications, due to the fast transients required to meet the road load and due to the size restrictions inside a vehicle. However, for stationary applications, these issues are not nearly as challenging, because a stationary engine runs predominantly at a constant speed and the load changes relatively slowly. Under these conditions, combustion control becomes much more tractable. External components (e.g. a burner and heat exchanger for starting the engine [9]) can easily be installed in stationary engines, since size restrictions are typically not as strict as for transportation applications.

This paper presents an analytical evaluation of the applicability of Homogeneous Charge Compression Ignition Engines (HCCI) for small-scale cogeneration (less than 1 MWe) in comparison to five previously analyzed prime movers. The five comparator prime movers include stoichiometric SI engines, lean burn SI engines, diesel engines, microturbines and fuel cells.

The parameters used in the analysis are listed in Table 1. We assume no heat losses or pressure drop in ducts for the different fluids. Water is used as the heat transfer fluid. All prime movers run on natural gas, except for the diesel engine. The stoichiometric SI engine operates with exhaust gas recirculation (EGR). The lean burn SI engine, the diesel engine, the microturbine and the HCCI engine operate with air dilution. The engines run turbocharged at 3 bar absolute intake. The gas turbine runs at 10:1 pressure ratio across the compressor and turbine. The SI engines have a low (10:1) compression ratio due to the limitations caused by knock. An upper limit is set on the water temperature leaving the SI engines (100°C) due to the possibility of knock if the engine overheats. The HCCI engine and the diesel engine can operate at a higher compression ratio (15:1) and at higher water temperature, since they are not limited by knock. The fuel heating power input is set at 600 kW, to correspond to an electric power output of approximately 200 kW.

Five of the six systems being considered (engines and microturbine) are analyzed with thermodynamic models. Fuel cells are evaluated based on information given in a recent publication [5]. The models use an iterative equation solver [10] and computational property tables to determine all conditions along the process. The conventional definitions of heat exchanger effectiveness [11] and turbine and compressor isentropic efficiency [12] are applied to the analysis. Engine brake thermal efficiency is calculated based on a thermodynamic engine model [3] that includes a heat transfer model [13] and a friction model [14]. The model takes into account the composition, pressure and temperature of the intake gases, along with the burn duration (10-90% of chemical heat release), which was set at 30 crank angle degrees. For the HCCI engine, a chemical kinetics model [15] was used to determine ignition timing and efficiency. Engine efficiency calculations assume a displacement of 2 liters per cylinder, typical of the 200-kW engine class. For the gas turbine system, the efficiency is determined directly from the thermodynamic model assuming appropriate isentropic efficiencies for the compressor and the turbine.

Once all system parameters are set, the equation solver determines the water flow rate that can be maintained while meeting the outlet temperature requirement. Finally, the heating power is calculated by finding the change in water enthalpy between the return and delivery conditions and multiplying this by the water mass flow rate. For simplification in this screening analysis, it is assumed that all of the electricity and recovered heat can be utilized. In an actual system, there will invariably be load factors that will impact the ultimate recovery and use. These will depend on the specific load being met, any staging of prime movers, and the thermal storage in the system.

The analysis considers two cases. Case 1 assumes that the prime mover is integrated to a facility that simultaneously requires power and heating. In Case 2, the cogeneration facility requires power and chilling. The next sections describe the analysis procedure, the systems being analyzed and the results for system efficiency and costs.

Case 1: Combined Power and Heating Applications

For Case 1 we assume that the return water enters the engine at 80°C and the hot water delivery temperature is 120°C (except for the fuel cell, where delivery temperature is 160°C [5]). These temperatures are considered typical of cogeneration facilities [16].

The Stoichiometric Spark-Ignited Engine: The stoichiometric SI cogeneration system for Case 1 is shown in Figure 1. The air and fuel mixture is first run through a compressor, where the fresh charge is pressurized to 3 bar absolute while simultaneously heating up considerably. At this point, the fresh charge is mixed with the residual gases, further increasing the charge temperature (up to 169°C). These gases need to be cooled (to about 60°C) to avoid knock and to increase the volumetric efficiency. Due to the low target temperature, the cogeneration water cannot be used for cooling the charge. Instead, additional water at near ambient temperature needs to be used. This presents two problems. First, the thermal energy of the intake charge (32 kW) is wasted. Second, supplying the cold water requires either a stream of continuous domestic water or a cooling tower. Both of these options are expensive [17].

Knock also limits the maximum water jacket temperature that can be used, and this is set at 100°C. After leaving the engine, the water flows through a heat exchanger where it recovers heat from the exhaust gases. The water finally flows past the EGR cooler before being directed to satisfy the heating load. As Figure 1 shows, part of the 100°C water flowing out of the engine cannot be used in the system, because the thermal energy available in the exhaust is not enough to raise the temperature of all the water to the delivery temperature (120°C). Allowing the engine to run with hotter water would permit a better utilization of available thermal energy. However, this may also result in knock, and it is therefore not acceptable.

The engine compression ratio is limited by knock at 10:1 (see, for example [1]). The lower compression ratio reduces the electric efficiency of the system. The engine brake efficiency is calculated at 34.2%, and the heating efficiency is 41.9% (251 kW/600 kW). Considering a 95% generator efficiency, the electric efficiency is 32.5% and the total fuel utilization efficiency is 74.4% (32.5%+41.9%).

Finally, it is worth noting that the intake temperature (and therefore the heating efficiency) in a SI engine can be increased at the expense of reducing the electric efficiency of the system. For example, in a recent experiment [18], the intake temperature was increased from 50°C to 90°C. As the charge got hotter, the ignition timing had to be retarded to avoid knock. Retarding ignition reduced the engine efficiency by 1.5%.

<u>The Lean Burn Spark-Ignited Engine:</u> A schematic of the cogeneration system with a lean burn SI engine for combined power and heating is shown in Figure 2. The fuel-air mixture is first compressed to 3 bar absolute and then it is run through an intercooler. The intercooler reduces the intake temperature to avoid knock and to increase the volumetric efficiency of the engine. As in the stoichiometric SI engine, heat transfer in the intercooler (33 kW) is wasted, since it occurs at a temperature too low for cogeneration applications. Using the intercooler also implies the additional expense of having a source of cold water or a cooling tower.

For the lean burn SI engine the brake thermal efficiency is calculated at 38%, in good agreement with a recent publication [19]. Heat recovery occurs in the engine and through a heat exchanger between the water and the exhaust gases. The system recovers 207 kW of heating power, for a heating efficiency of 34.5%. The fuel utilization efficiency is 70.6%.

While the lean SI engine delivers reasonable electric and heating efficiencies, NO_x emissions present the biggest challenge to implementation of these engines. In this paper, we consider that the lean SI engine produces 0.6 g/kWh of NO_x, which was recently determined to be the minimum possible NO_x emissions that can be produced in a lean burn SI engine without aftertreatment [4]. This is significantly higher than the 2007 CARB NOx standards (0.032 g/kWh). As previously discussed, lean NO_x aftertreatment not economical for small facilities. Without successful NO_x control, lean burn SI engines cannot be considered a viable option for small-scale CHP systems in areas with strict emissions standards.

The Diesel Engine: The inlet air into a diesel engine is first compressed to 3 bar absolute (Figure 3), which raises the air temperature to 153°C. At this point, it is possible to cool the air to improve volumetric efficiency (diesel engines are not limited by knock). However, for a cogeneration application, it is best not to cool the intake air to increase the heating efficiency of the system and to avoid the need of cooling water in the intercooler. Using hot intake without intercooling reduces the volumetric efficiency and the specific power output from the engine, but this is considered a good trade-off due to the improved heating efficiency of the system.

The engine brake thermal efficiency is calculated at 41.8%. The heating water loop goes from the engine to a heat exchanger being heated by the exhaust gases. Diesel engines are not restricted to run with low water temperatures, and therefore no hot water from the engine needs to be wasted. The heating efficiency is quite high (49%), and the overall fuel utilization efficiency is 88.7%. While these efficiencies are very high, the outstanding issue is the very high NOx emissions from diesel engines that will limit them to regions without strict emissions standards.

The Microturbine: The microturbine cogeneration system for Case 1 is shown in Figure 4. It is assumed that the intake air is compressed to 10 bar before flowing into a combustor. The maximum equivalence ratio is limited by the maximum temperature that the turbine blades can withstand. Assuming 1500°C as an upper limit yields a 26.0% mechanical efficiency, in good agreement with a recent publication [20].

Microturbines have a low mechanical efficiency, but they do have a good potential for CHP applications due to the high temperature of the exhaust gases (924°C). The model shows that it is possible to recover 335 kW of the available fuel energy, for a heating efficiency of 56%. The fuel utilization efficiency is 80.6%.

Microturbines produce approximately 0.16 g/kWh of NO_x [21]. While this is a very low value, it is higher than the CARB 2007 standards. Microturbine cost ($1400/kW) is also an issue in most applications

The Fuel Cell: Fuel cells have been evaluated based on information given in a recent publication for phosphoric acid fuel cells [5]. The fuel cell has 40% electric efficiency. In addition to this, 17% of the energy input into the fuel cell is recovered at high temperature (160°C) from the cell stack and 21% of the energy can be recovered from the exhaust gases at a low temperature (60°C). The temperature of the energy recovery from the exhaust gases (60°C) is too low for the application specified here, so the analysis only considers the 17% of the energy that can be recovered at a high temperature.

The Homogeneous Charge Compression Ignition (HCCI) Engine: The HCCI engine runs very lean (equivalence ratio=0.4) to keep the combustion temperature low, avoiding NO_x emissions and keeping the peak cylinder pressure and the rate of pressure rise to manageable levels. The schematic of the HCCI cogeneration system for Case 1 is shown in Figure 5. The intake air is compressed to 3 bar absolute before entering the engine.

HCCI engines are not limited by knock, and therefore there is no need for an intercooler, although not having an intercooler decreases the volumetric efficiency and the specific power output.

The thermodynamic analysis of the HCCI engine shows 238 kW of mechanical power output (39.6% efficiency). The efficiency is lower than for the diesel engine due to the lower power output, which magnifies the effect of friction losses on efficiency [22]. The heating power is 295 kW, corresponding to a 49.2% heating efficiency. While the fuel utilization efficiency (86.8%) is slightly lower than for a diesel engine, it appears that the much lower NO_x emissions in HCCI engines may make HCCI engines the best choice for cogeneration in places where the NO_x standards are very strict. The combination of over 40% brake thermal efficiency and NO_x emissions that comply with the CARB 2007 standards has been demonstrated in a recent HCCI engine experiment [23].

HCCI engines typically have high emissions of hydrocarbon and carbon monoxide. However, these emissions can be controlled with mature existing technology (oxidizing catalysts) to a level well within the CARB 2007 standards.

Case 2: Combined Power and Chilling Applications

For Case 2, the prime mover and heat recovery system are integrated into an absorption chiller. For the engines, we assume that chilling is accomplished with a single effect absorption unit operating at 120°C inlet and 115°C outlet temperature. These temperatures typically result in good performance for single effect absorption chillers [24]. Lower inlet and/or return chiller temperatures can be used, but only at the expense of considerably reduced specific power output, which results in a substantial increase in chiller capital cost [25]. On the other hand, the coefficient of performance (COP) for a single effect chiller is fairly insensitive to the operating temperatures, and is approximately equal to 0.7.

For the fuel cell and the microturbine, the exhaust temperature is high enough that it is possible to use a double effect chiller. We assume that the double effect chiller operates between 160°C and 155°C [24], and has a higher COP (equal to 1.1).

The operating conditions for the prime movers for Cases 1 and 2 are basically identical, since the temperature differences between the two cases affect the heat recovery system only and not the prime mover. Therefore, prime mover parameters such as electric efficiency and emissions are the same in Case 1 and Case 2. To avoid repetition, the system descriptions presented next mainly highlight differences between the two cases and do not go into system details that have been previously discussed for Case 1.

<u>The Stoichiometric Spark-Ignited Engine:</u> The stoichiometric SI cogeneration system for Case 2 is shown in Figure 6. For chilling, the return water temperature into the engine is at 115°C and it is therefore too hot to be circulated through the engine, which can run

with a maximum water temperature of 100°C. Therefore, the engine cooling water cannot be used for running the chiller, and its thermal energy is wasted, along with the thermal energy rejected in the intercooler. Only the exhaust gases can be used as a source of heat for the chilling system. Consequently, the heating efficiency of the stoichiometric SI engine for Case 2, 20.6%, is considerably lower than for Case 1 (41.9%). The chilling efficiency is 14.4% and the fuel utilization efficiency is 46.9%.

The Lean Burn Spark-Ignited Engine: Figure 7 shows the cogeneration system for a lean burn SI engine for combined power and chilling. The lean burn SI engine has the same 100**°C** upper limit in water temperature as the stoichiometric SI engine, and the engine coolant energy has to be wasted once again due to its low temperature. The resulting heating efficiency of the lean burn SI engine for combined power and chilling is only 17.2% and the chilling efficiency is 12%.

The Diesel Engine: Diesel engines can operate with a high water temperature, so they can accept the hot water returning from the absorption chiller at 115°C (Figure 8). Diesel engines can therefore provide high heating efficiency for operation of an absorption chiller. The heating efficiency for Case 2 is 48.5%, almost identical to the heating efficiency for Case 1 (49%).

The Microturbine: The microturbine cogeneration system for Case 2 is shown in Figure 9. The high temperature exhaust in the microturbine makes it possible to use a double effect absorption chiller with a high (1.1) COP. The chilling efficiency of the microturbine is 56.5%, and the fuel utilization efficiency is 81.2%, highest of all systems considered for Case 2.

The Fuel Cell: Phosphoric acid fuel cells have a high enough coolant temperature (160°C) that makes it possible to combine them with a double effect chiller. The heating efficiency of the fuel cell is 17%, and the chilling efficiency is 18.7%.

The Homogeneous Charge Compression Ignition (HCCI) Engine: The HCCI engine-based cogeneration system for combined power and chilling (Figure 10) can achieve a high heating efficiency (48.4%) because it can run with high inlet water temperature (115**°C)**. The heating efficiency is very similar to the heating efficiency for Case 1 (49.2%). The fuel utilization efficiency is 71.5%.

Summary of Results

The results of the analysis are summarized in Table 2. Table 2 includes efficiencies, fuel savings and costs. Efficiencies and fuel savings are considered more accurate than costs, due to the uncertainty and variability of costs. Capital costs listed in Table 2 include the prime mover and generator only. Costs of heat exchangers, cooling tower, and other accessories are not included. Operation and maintenance costs are not included in the analysis due to the high uncertainty in these values, but it is considered that the spark-ignited engines require more maintenance than other prime movers due to the need to replace spark plugs and the 3-way catalytic converter in the stoichiometric SI engine.

Table 2 is divided in three parts. The first part of the table includes results (mainly engine performance, cost and emissions) that apply to both Case 1 (combined power and heating) and Case 2 (combined power and chilling). This is followed by parts that apply exclusively to Case 1 and then to Case 2.

Table 2 shows that the fuel cell has the highest electric efficiency (40%), followed closely by the diesel engine. HCCI engines have much lower specific power (or brake mean effective pressure, BMEP) than other engines due to the high level of dilution and the high intake temperature. Lower BMEP results in bigger displacement for a given power output and therefore higher capital cost. Spark-ignited engines and diesel engines have reasonably small displacements and capital costs due to their high BMEP[3]. Fuel cells and microturbines are considerably more expensive than the engines.

Emissions of NO_x for the stoichiometric SI engine with a 3-way catalyst are obtained from a recent publication [17]. For the lean burn SI engine and the diesel engine we assume the minimum viable emissions established in a recent publication [4]. Emissions for microturbines are based on recent experience on existing commercial models [21]. For the HCCI engine, we use recent experimental values [23]. Emissions of other pollutants (HC, CO, PM) are not listed in Table 2. We consider that all prime movers can meet emissions standards for these pollutants with available technology (oxidizing catalyst and a particulate matter filter).

The results for Case 1 in Table 2 indicate that the diesel engine has the highest fuel utilization efficiency (88.7%) followed closely by the HCCI engine (86.8%). Microturbines have very high heating efficiency due to the availability of high temperature exhaust, and fuel cells have a high electric efficiency but a low heating efficiency due to the low temperature of the exhaust gases that cannot be used to meet the demand specified in this paper.

All the options for the cogeneration systems for Case 1 result in considerable energy savings with respect to a case with no cogeneration, in which it is assumed that the electricity is generated from natural gas in a power plant with a 40% generation and distribution efficiency and the heat is generated from natural gas with a 90% efficient burner [26]. The diesel engine shows the highest savings in energy, consuming only 65% of the energy of the case with no cogeneration. The HCCI is a close second, consuming 68% of the energy consumed without cogeneration.

For Case 1, all the prime movers operating on natural gas reduce the cost of fuel with respect to the case without cogeneration. However, diesel fuel is considerably more expensive than natural gas ($0.040/kWh for diesel vs. $0.017/kWh for natural gas).

[3] The BMEP given in Table 2 for the stoichiometric spark ignited engine with EGR (22.4 bar) has not been demonstrated in a commercial product due to durability issues. However, there are no fundamental reasons that keep the stoichiometric SI engines from achieving these high BMEP levels.

Therefore, the fuel consumed by the diesel engine has a higher cost (by $ 74000/year) than the natural gas consumed without cogeneration, even though the diesel engine has the highest fuel utilization efficiency of all systems being considered. The HCCI engines result in the lowest cost of fuel for operation during one year, with a savings of $44,000 with respect to a case with no cogeneration. SI engines save $25,000-26,000 of fuel per year. Microturbines save $21,000 and fuel cells save $17,000.

For combined power and chilling (Case 2), the microturbine achieves an extremely high chilling efficiency (56.5%) due to the availability of high temperature exhaust that allows the use of a high efficiency double effect chiller. The fuel utilization efficiency for the microturbine is 80.6%. Diesel and HCCI engines have high fuel utilization efficiency for chilling because they can run with high cooling water temperature (120°C). The thermal energy of the cooling water can therefore be used, along with the thermal energy of the exhaust, to run the absorption chiller. SI engines cannot run at high water temperatures due to knock limitations, and therefore only the thermal energy of the exhaust can be used for running the absorption chiller, resulting in low chilling and fuel utilization efficiencies.

All systems except the stoichiometric SI engine system save energy relative to the base non co-generation chilling system considered: one wherein the electricity is obtained from the grid the same as for the heating comparison system and the chilling is provided by an electrically driven mechanical chiller with a COP of 2.0. The low chilling efficiency of the stoichiometric SI engine and absorption chiller results in it having greater energy consumption than with no cogeneration. This also results in a negative cost savings for this system. All other systems have cost savings relative to the no cogeneration system except for the diesel engine system which again has a much greater cost of operation due to the high relative cost of the diesel fuel. The HCCI engine saves the most money ($33,000 per year), and the microturbine is second ($21,000 per year).

Conclusions

This paper has presented an analysis of the applicability of six different prime movers for small-scale cogeneration (less than 1 MWe). Five of these have been previously analyzed, including stoichiometric SI engines, lean burn SI engines, diesel engines, microturbines and fuel cells. The sixth option, HCCI engines, is a relatively new type of engine that has some fundamental differences with respect to other prime movers. These differences can be exploited to obtain highly efficient, clean and inexpensive cogeneration systems. Two cases are analyzed. In Case 1, the cogeneration facility requires combined power and heating. In Case 2, the requirement is for power and chilling. The main results of this paper can be summarized as follows.

- The stoichiometric SI engine has good fuel utilization efficiency for combined power and heating (74.4%), very low emissions and high power density. The fuel utilization efficiency is considerably lower for combined power and chilling

(46.9%) due to the limitation in water temperature to avoid knock. Stoichiometric SI engines require high maintenance and have a low electric efficiency.

- The lean burn SI engine has a good electric efficiency and fuel utilization efficiency (70.6%) for combined power and heating. For combined power and chilling the fuel utilization efficiency is considerably lower (48.1%). Lean burn SI engines have high emissions of NO_x and do not meet current CARB standards. This may restrict their use in urban areas where strict emissions standards may exist.
- Diesel engines are very efficient for cogeneration, with a fuel utilization efficiency of 88.7% for Case 1 and 73.7% for Case 2. However, diesel engines have high emissions of NO_x that restrict their applicability in urban areas. In addition to this, diesel fuel is considerably more expensive than natural gas. Therefore, the cost of fuel consumed by the diesel engine is higher than the cost of natural gas necessary to provide the electric and heating loads if no cogeneration system is installed.
- Microturbines have very high heating efficiency. The high heating efficiency and the high temperature of the waste heat makes them the system with the highest fuel utilization efficiency for combined power and chilling (80.6%). However, they are expensive, have a very low electric efficiency and have NO_x emissions that, although low, do not meet the oncoming CARB 2007 standards.
- Fuel cells have a high electric efficiency and extremely low emissions. Their heating efficiency is quite low. With their high electric efficiency, their low heating efficiency and their very high cost they are limited to facilities that require very high power quality since they do deliver higher quality power than the typical installation of the other prime movers.
- HCCI engines closely approach the fuel utilization efficiency of diesel engines without the high emissions of NO_x and the expensive fuel. HCCI engines offer a new alternative for cogeneration that provides a unique combination of low cost, high efficiency, low emissions and flexibility in operating temperatures that can be optimally tuned for cogeneration systems. HCCI engines are the most efficient technology that meets the oncoming 2007 CARB NO_x standards for cogeneration engines. The HCCI engine appears to be a good option for cogeneration systems and merits more detailed analysis and experimental demonstration.

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Table 1. Parameters used in the cogeneration analysis. Case 1 is combined power and heating and Case 2 is combined power and chilling.

parameter	**SI engine stoich.**	**SI engine lean**	**Diesel engine**	**Micro-turbine**	**Fuel cell**	**HCCI engine**
fuel	Natural gas	Natural gas	Diesel	Natural gas	Natural gas	Natural gas
Equivalence ratio	1.0	0.7	0.7	0.5	1.0	0.4
EGR Fraction (exhaust gas recirculation)	0.2	0	0	0	0	0
Intake manifold pressure	3 bar absolute	3 bar absolute	3 bar absolute	10 bar into turbine	1 bar absolute	3 bar absolute
Exhaust aftertreatment	3-way catalyst	Oxy catalyst	none	none	none	Oxy catalyst
Compression ratio	10	10	15	-	-	15
Fuel heating power into engine	600 kW	600 kW	600 kW	600 kW	600 kW	600 kW
Maximum hot water temperature out of engine	100°C	100°C	-	-	-	-
Return water temperature into engine, Case 1	80°C	80°C	80°C	80°C	80°C	80°C
Hot water delivery temperature, Case 1	120°C	120°C	120°C	120°C	160°C	120°C
Return water temperature into engine, Case 2	115°C	115°C	115°C	155°C	155°C	115°C
Hot water delivery temperature, Case 2	120°C	120°C	120°C	160°C	160°C	120°C

Table 2. Summary of results for analysis of the six systems being considered for combined heating and power.

parameter	**SI engine stoich.**	**SI engine lean**	**Diesel engine**	**Micro-turbine**	**Fuel cell**	**HCCI engine**
Results that apply for both Case 1 and Case 2						
Brake thermal efficiency, %	34.2	38.0	41.8	26.0	-	39.6
Electric efficiency, %	32.5	36.1	39.7	24.7	40.0	37.6
Brake mean effective pressure (BMEP), Bar	22.4	23.2	19.4	-	-	10.8
Engine displacement (liters)	6.1	6.5	8.58	-	-	14.6
Capital cost[1],$/kWe	323	316	357	1400	3000	562
Total capital cost[2], $	63,000	68,400	85,037	207,000	720,000	135,000
NO_x emissions (g/kWh)	0.02	0.6	1.3	0.16	0	.014
Cogeneration system energy consumed in a year of operation[3], kWh	5.3×10^6	5.3×10^6	5.3×10^6	5.3×10^6	5.3×10^6	5.3×10^6
Results that apply to Case 1, combined power and heating						
Heating efficiency, %	41.9	34.5	49.0	55.9	17.0	49.2
Fuel utilization efficiency[4], %	74.4	70.6	88.7	80.6	57.0	86.8
Energy consumed with no cogeneration[5] in a year of operation, kWh	6.7×10^6	6.8×10^6	8.1×10^6	6.5×10^6	6.2×10^6	7.8×10^6
Energy savings in a year of operation[6], kWh	1.4×10^6	1.5×10^6	2.8×10^6	1.2×10^6	0.9×10^6	2.5×10^6
Cost savings in a year of operation[7], $	25,000	26,000	-74,000	21,000	17,000	44,000
Results that apply to Case 2, combined power and chilling						
Type of absorption chiller	single effect	single effect	single effect	double effect	double effect	single effect
Chiller COP	0.7	0.7	0.7	1.1	1.1	0.7
Heating efficiency, %	20.6	17.2	48.5	51.4	17.0	48.4
Chilling efficiency, %	14.4	12.0	34.0	56.5	18.7	33.9
Fuel utilization efficiency[8], %	46.9	48.1	73.7	81.2	43.4	71.5
Energy consumed with no cogeneration[9] in a year of operation, kWh	5.2×10^6	5.5×10^6	7.4×10^6	7.0×10^6	6.5×10^6	7.2×10^6
Energy savings in a year of operation[6,] kWh	-3.8×10^4	2.8×10^5	2.1×10^6	1.7×10^6	1.2×10^6	1.9×10^6
Cost savings in a year of operation[7], $	-650	4700	-84,000	29,000	21,000	33,000

Notes to Table 2.

1. For engine and generators calculated as \$100/kW for the generator and \$500/kW for an engine with a BMEP=10. The generator cost is constant for all cases and the engine cost is scaled according to the BMEP. Capital cost includes prime mover only, no heat exchangers or aftertreatment system.
2. Calculated as the cost per kWe times the electric power output.
3. Calculated as 600 kW times 365x24 hours in a year.
4. For Case 1, fuel utilization efficiency is defined as electric efficiency plus heating efficiency.
5. Energy consumed if there is no cogeneration system, assuming 40% efficiency for generation and distribution of electricity and 90% heater efficiency [26].
6. Difference between energy consumption without cogeneration system and energy consumption with cogeneration system.
7. Difference between the cost of energy consumed without cogeneration system and the cost of energy consumed with cogeneration system, assuming natural gas cost at \$0.017/kWh (\$5/MMBtu) and diesel fuel cost at \$0.040/kWh (\$1.5/gal)
8. For Case 2, fuel utilization efficiency is defined as electric efficiency plus chilling efficiency.
9. Energy consumed if there is no cogeneration system, assuming 40% efficiency for generation and distribution of electricity and assuming an electric vapor compression chiller with COP=2.0.

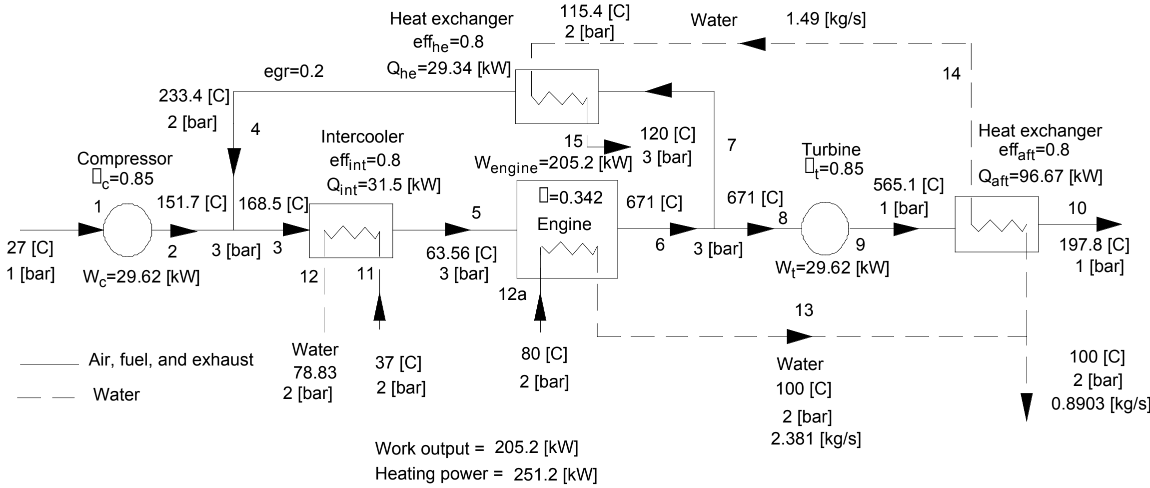

Figure 1. Schematic of stoichiometric SI engine cogeneration system for Case 1, power and heating generation.

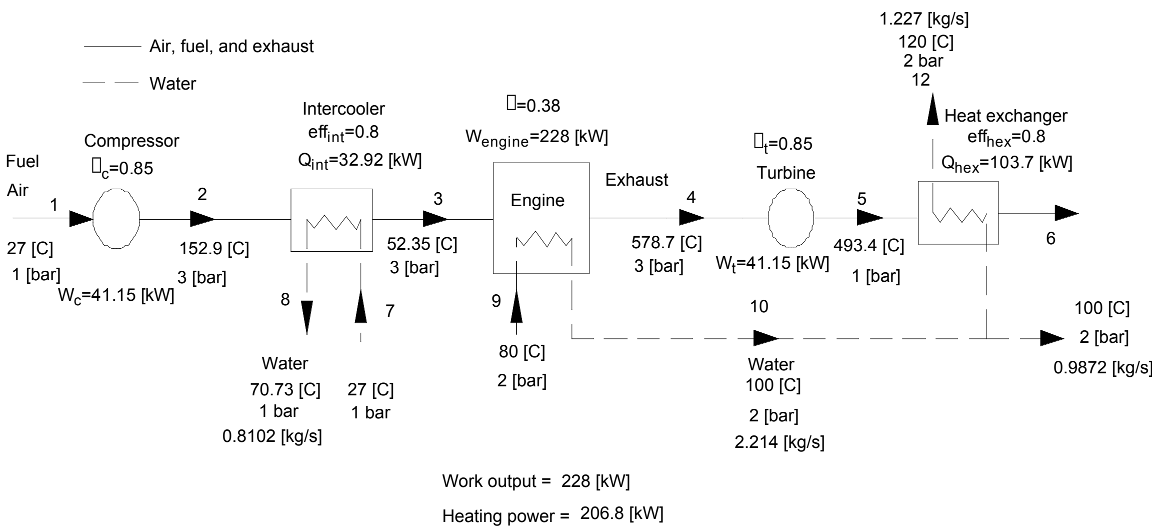

Figure 2. Schematic of lean burn SI engine cogeneration system for Case 1, power and heating generation.

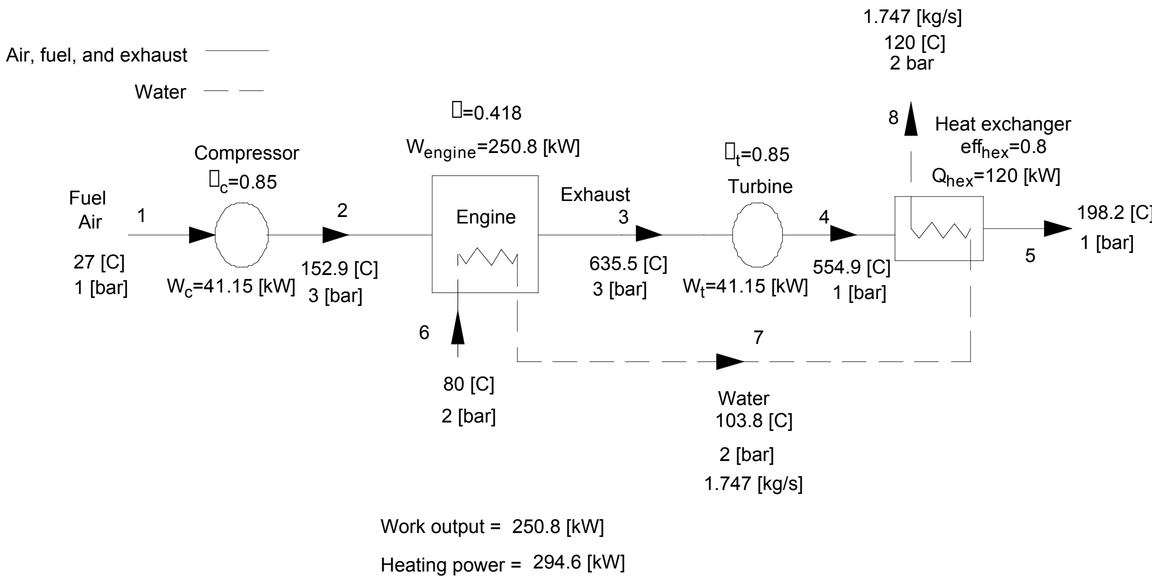

Figure 3. Schematic of diesel engine cogeneration system for Case 1, power and heating generation.

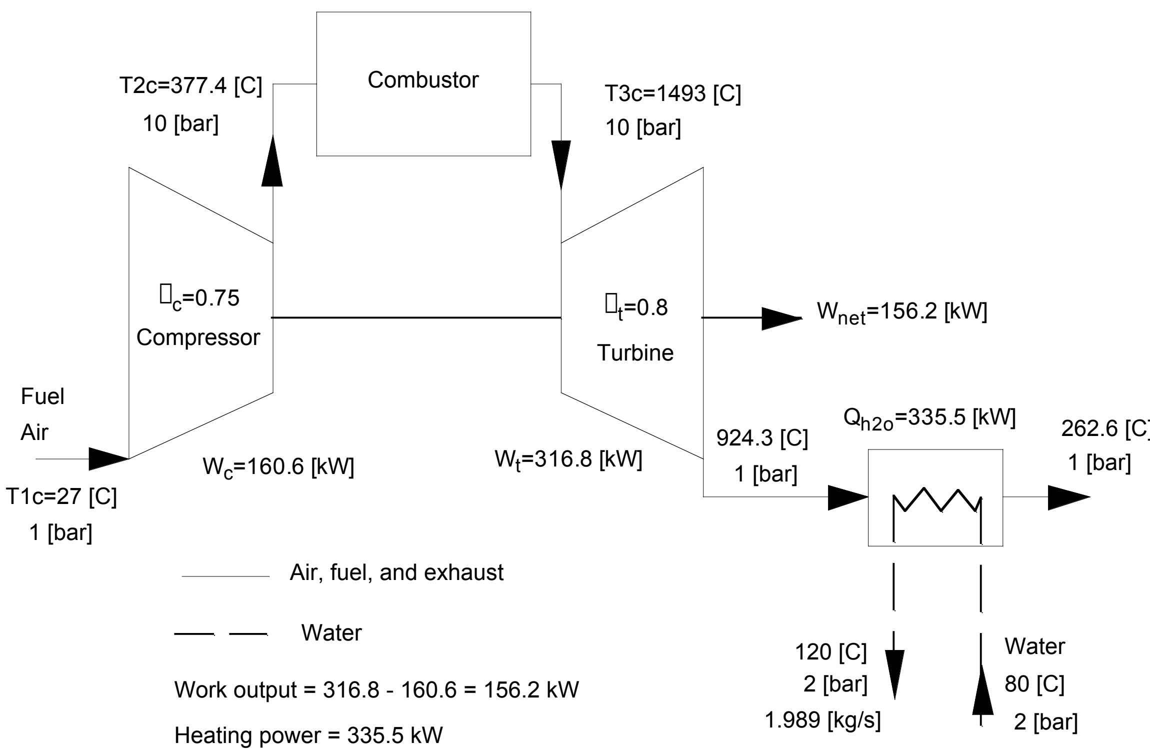

Figure 4. Schematic of microturbine cogeneration system for Case 1, power and heating generation.

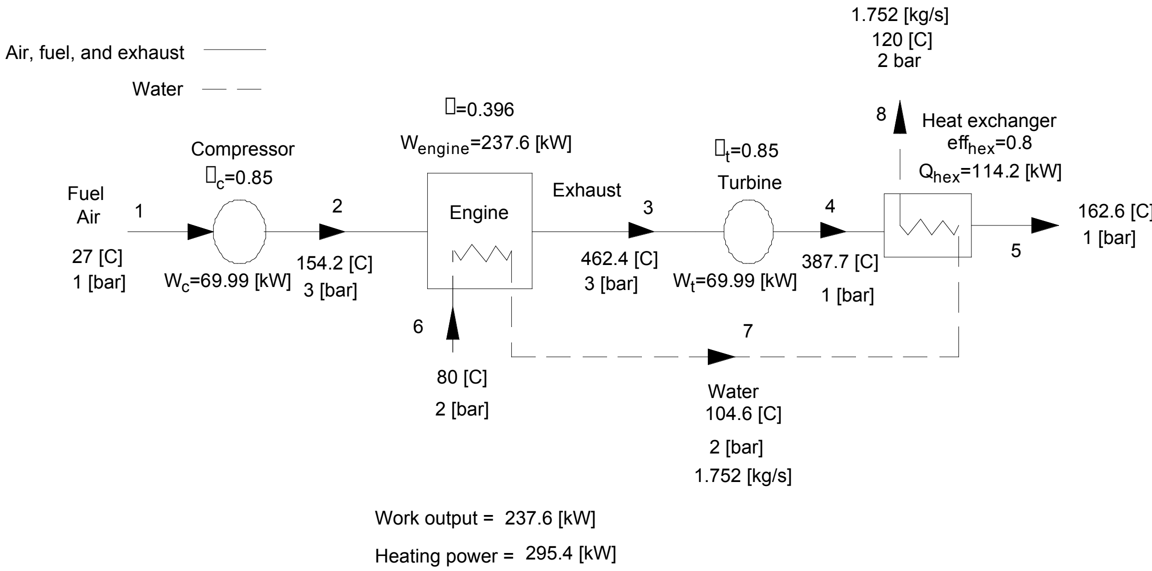

Figure 5. Schematic of HCCI engine cogeneration system for Case 1, power and heating generation.

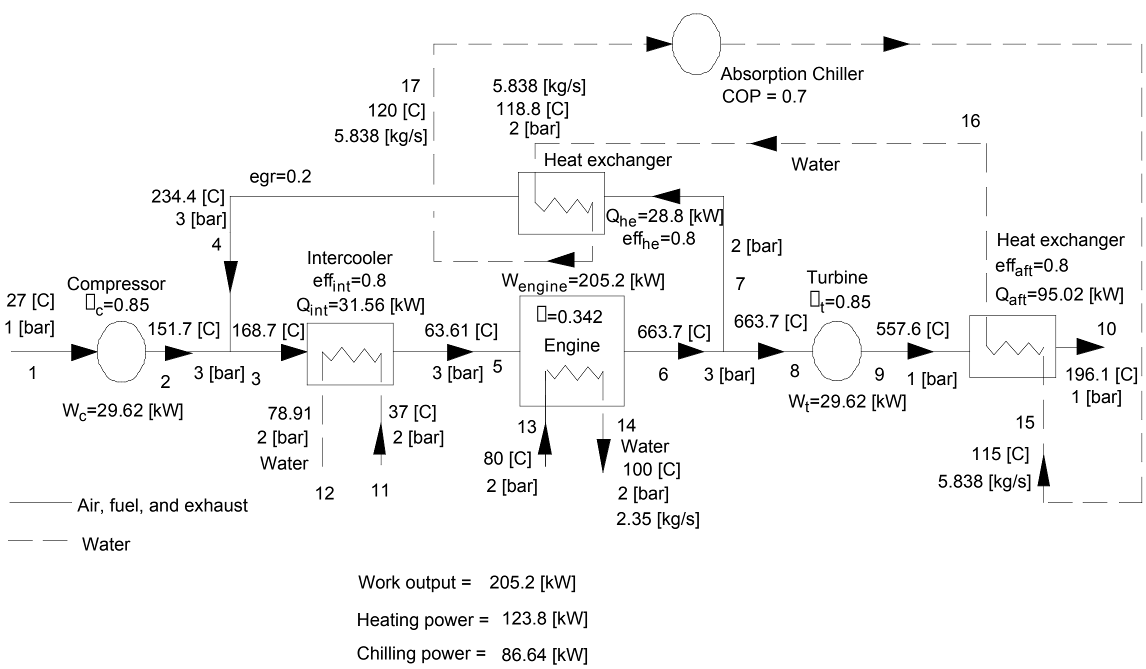

Figure 6. Schematic of stoichiometric SI engine cogeneration system for Case 2, power generation and chilling.

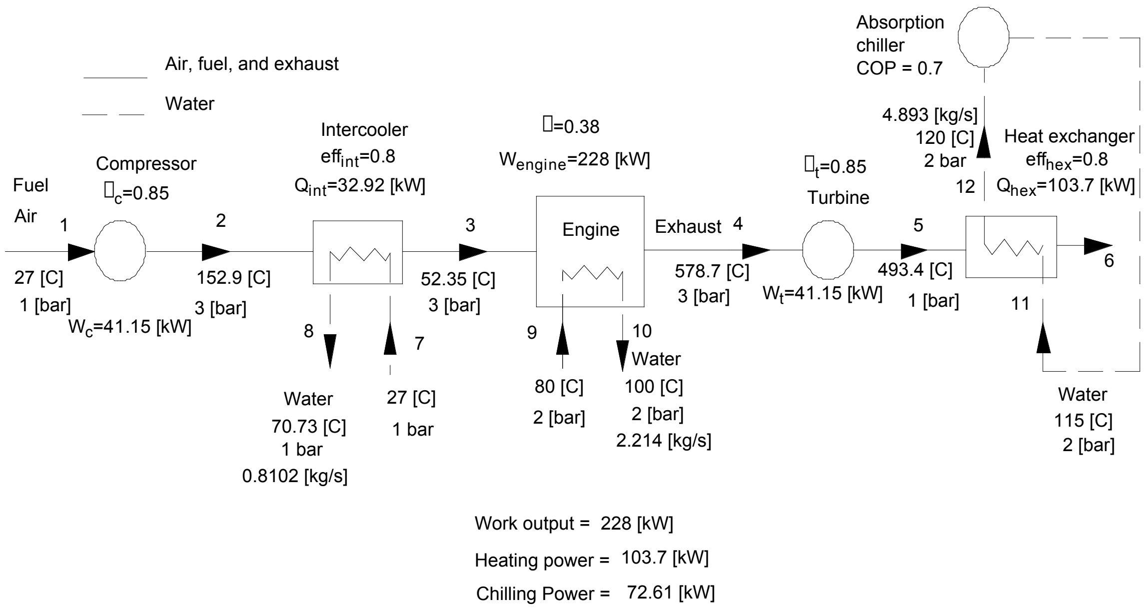

Figure 7. Schematic of lean burn SI engine cogeneration system for Case 2, power generation and chilling.

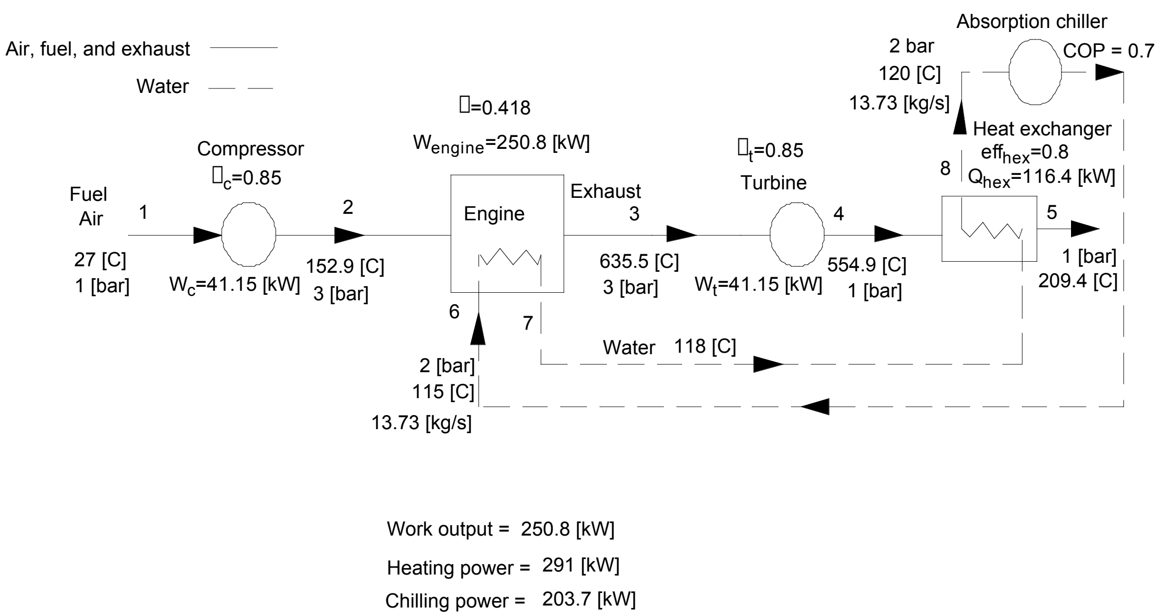

Figure 8. Schematic of diesel engine cogeneration system for Case 2, power generation and chilling.

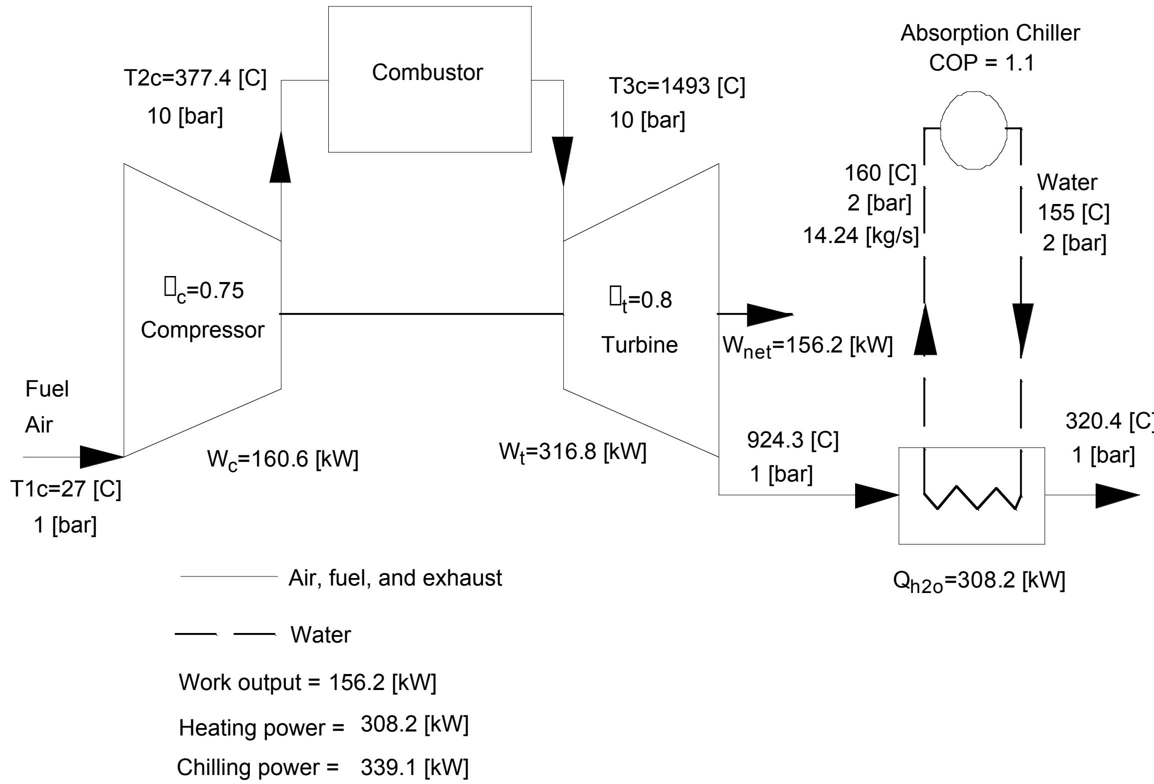

Figure 9. Schematic of microturbine cogeneration system for Case 2, power generation and chilling.

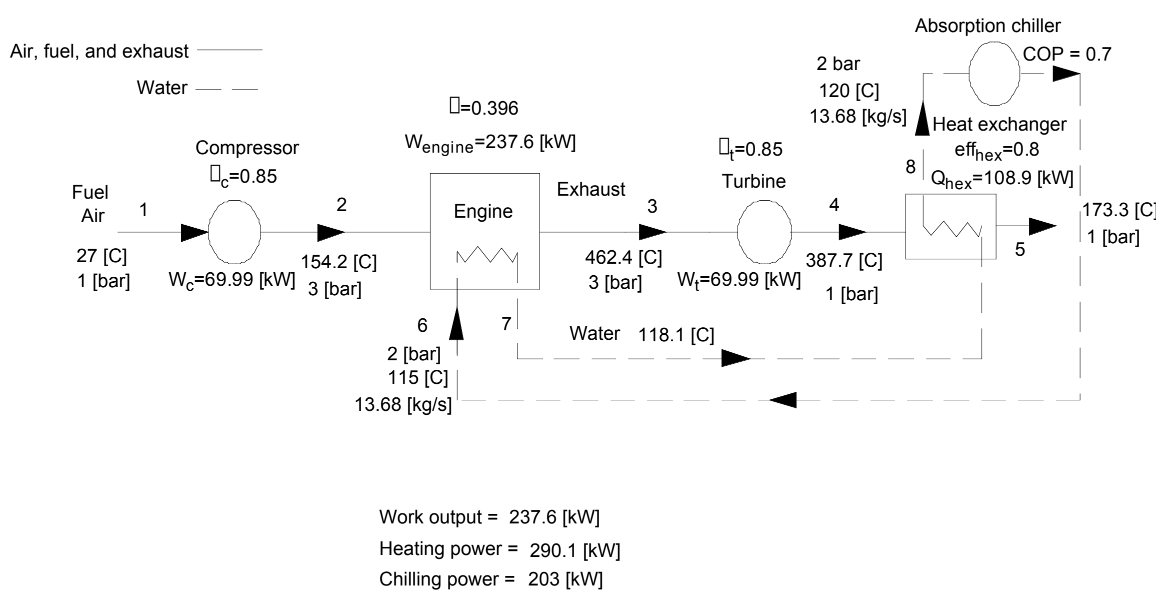

Figure 10. Schematic of HCCI engine cogeneration system for Case 2, power generation and chilling.